REMARKS

Entry of the present amendment and reconsideration of the claims is respectfully requested.

I. Status of the Claims

Claims 21 and 29 have been amended and the amendments do not add new matter.

Claims 21-25 and 27-45 are pending in the application.

Claim 29 is objected to for informal matters. Applicant has amended the claim to address the informalities and respectfully request that the objection be withdrawn.

II. Acknowledgment of Allowable Subject Matter

Applicant would like to thank the Examiner for the acknowledgment of allowable subject matter in claims 28, 29, 37-42 and 44-45.

III. Telephone Interview

Applicant thanks the Examiner for discussing proposed amendments to the claims and the Crosby and Roche references in the telephone interview of March 29, 2004 with Applicant's representative, Louis DelJuidice. No agreement was reached regarding the application.

IV. Rejections under 35 U.S.C. § 101 - Double Patenting

Claims 21-25, 27, 31-36 and 43 have been rejected under 35 U.S.C. §101 for nonstatutory judicially created doctrine of obviousness-type double patenting over claim 3 of U.S. Patent No. 6,116,138 to Achten in view of U.S. Patent No. 2,550,405 to Crosby and U.S. Patent No. 5,251,442

to Roche. Achten's invention relates to a hydraulic transformer that controls a hydromotor without any additional valves or controls. Achten discloses a simple hydraulic design wherein each hydromotor has its own hydraulic transformer. Additionally, neither Crosby nor Roche, as argued below, can be properly combined with each other nor can they be combined with Achten to reject the claims.

In light of the amendments and the reasons set forth herein, Applicant respectfully requests withdrawal of the rejections under 35 U.S.C. § 101 and double-patenting and urge allowance of the claims as now amended.

V. Rejections under 35 U.S.C. § 103

Claims 21-25, 27 and 30-32 have been rejected under 35 U.S.C. § 103(a) as being unpatentable over U.S. Patent No. 2,550,405 to Crosby in view of U.S. Patent No. 5,251,442 to Roche. The Examiner states that Crosby discloses all the elements of the claimed invention except for a sensor that measures flow in the connecting line and that Roche discloses most of the elements of the claimed invention, including a flow sensor that measures flow in the line between the transformer and the hydromotor.

Applicant respectfully traverses the above rejection. Applicant has amended claim 21 to recite "the control means comprises a sensor for measuring the flow rate of the fluid flow in the connecting line between the one of the rotating and linear hydromotor and the hydraulic transformer and means for restricting the flow rate." In a hydraulic unit connected to a pressure source that supplies fluid at a high capacity or the hydraulic unit can take in hydraulic fluid at a high capacity from a low pressure line, there is a serious risk of injury if the load falls off or if the speed of the hydromotor changes in an unexpected way. Prior art transformers cannot change their setting

quickly enough so that the flow rate does not change uncontrolled. The presently claimed invention can quickly change the setting of the transformer to control the flow rate. The claimed control means both detects the flow rate of the fluid in the connecting line and can restrict the flow rate.

In contrast, the prior art of record does not acknowledge the dangers of an uncontrolled flow rate from the loss of a load or taking a temporary unlimited supply of high pressure fluid.

Applicant admits that Roche discloses a control system that monitors pressure and flow rate in the output line. *See*, e.g. Rosch, Figure 6. Based on the monitored pressure and flow rate, Roche's control system calculates new setting based on a "set point" for the system pressure and calculates new settings for the shutoff valves and the modulating valves in order to match the monitored flow rate with the flow rate demand of the load. However, contrary to the presently claimed invention, Roche teaches that "the control system provides for making large changes in flow rate to accommodate sudden changes in flow demand with little or no change to the system pressure." Roche, column 18, lines 39-42. Given the above, Roche teaches away from the present invention. Further, neither Roche nor Crosby teaches or suggests the problem solved by the present invention. Thus, one of ordinary skill in the art is not taught or motivated to combine Roche with Crosby to solve the problem of the present invention.

Additionally, 22-25, 27 and 30-32 depend from claim 21 and are allowable based at least on their dependency to the independent claim.

Applicant respectfully requests the present rejections be withdrawn.

For the Examiner's convenience, Applicant has attached hereto, as Exhibit A, the article *Cylinder Control with the Floating Cup Hydraulic Transformer*, The Eighth Scandinavian International Conference on Fluid Power, SICFP '03, presented May 7-9, 2003, Tampere Finland,

by Georges Vael, Peter Achten, and Jeroen Potma. The article outlines the state of the art prior to the present invention and outlines some of the uses and benefits of the present invention.

CONCLUSION

In view of the above, each of the presently pending claims in this application is believed to be in immediate condition for allowance. Accordingly, the Examiner is respectfully requested to pass this application to issue.

Dated: April 28, 2004

Respectfully submitted,

By______________________

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EXHIBIT A

The Eighth Scandinavian international Conference on Fluid Power, SICFP'03, May 7-9, 2003, Tampere, Finland

CYLINDER CONTROL WITH THE FLOATING CUP HYDRAULIC TRANSFORMER

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ABSTRACT

The Common Pressure Rail is a hydraulic drive line lay-out that holds great promise with respect to efficiency, controllability and modularity. In the field of mobile machinery, it can be a direct competitor to the current drive line of choice, the Load Sensing system. However, the Common Pressure Rail concept was never viable because it lacked a good solution to drive linear loads.

The Innas Hydraulic Transformer was developed to be such a solution. Recently, with the introduction of the Floating Cup displacement principle, the outlooks for a competitive design of the Innas Hydraulic Transformer have become a lot better. A Floating Cup Hydraulic Transformer prototype may be expected on short term.

Consequently, the activities on Common Pressure Rail systems have been intensified. This paper describes one of the aspects of the design of a Common Pressure Rail System: the way in which a differential cylinder can be connected to a Common Pressure Rail through an Innas Hydraulic Transformer. Four different options are introduced and compared.

KEYWORDS

Common Pressure Rail (CPR), Innas Hydraulic Transformer (IHT), Cylinder Control, Floating Cup (FC) Principle.

1 INTRODUCTION

As a linear force generating element, the hydraulic cylinder is unrivaled in power and force density. For this reason, it is still almost the only drive option for the implement functions in mobile machines. Most of the hydraulic cylinders in these machines control the movement of a flexible kinematic structure, e.g. the boom of an excavator. They have to be controlled to obtain synchronized movements, regardless of the load situation. In the past, this control was left to skilled and experienced operators. Over time, the demands on productivity grew and in response to that, the complexity and potential of mobile machines increased. The load on the operators increased but also the dependency of their skills.

In order to improve the controllability of mobile machinery, so called 'Load Sensing' (LS) systems were developed. In Europe and in the United States the LS system is the system type of choice for modern mobile machinery. A typical LS circuit for an excavator is given in figure 1. In this LS system, each control section contains an upstream

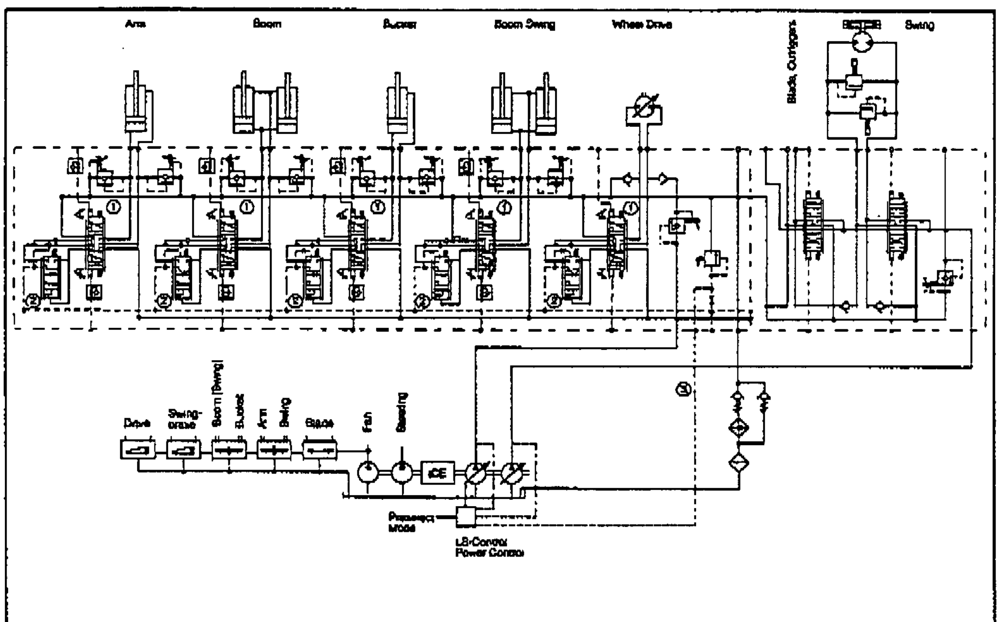

Figure 1: A typical LS excavator circuit

pressure balance, which realizes a constant pressure drop over the control valve. Consequently, the flow through the valve is a function of the actual valve position only, which is directly commanded by the operator. The flow – and with that the cylinder speed – has become independent of the actual downstream load pressure. Therefore, these systems are often said to have 'load independent flow control'.

Because all load pressures are sensed anyway, a logical next step is to compare them and use the highest value to control a supply pump, which gives a pressure level just above that value plus the required pressure drop over the valve. In this way the pump output pressure is restricted to the minimum value required for simultaneous movement of all cylinders. In early literature on LS systems, their energy saving capability is often stressed. This, however, is probably not the main reason for their success: the load independent flow control is.

In spite of these advantages, the LS system type also has its drawbacks. It takes careful tuning to get an LS system to work properly. Consequently it is not well suited for a really modular system approach. Although the LS system offers the possibility to control the pump pressure to a limited value above the highest load, unnecessary power losses are still suffered at the loads that do not require this pressure.

A system type that holds the same potential for precise control and synchronization of the cylinders is the 'Common Pressure Rail' (CPR) system. In this system type, a high pressure rail (typically 30 MPa) and a low pressure rail (typically 1 MPa), run through the whole vehicle. A pressure controlled pump is used to maintain the high pressure level and accumulators are attached to the high and the low pressure rail. The accumulators help to keep the rail pressure semi-constant and allow a temporary storage of energy. In this way the loads can recuperate energy to the rail, which implies a large efficiency gain for a lot of mobile machines. The pump is set up to deliver only the average power requirement, which means that a smaller engine and pump can be installed.

All loads in the CPR system operate from the same pressure and are controlled directly at the load. In this way they do not influence each other, as long as the pump is able to deliver the required average power to the rail. This is a distinct and important difference to the LS system type, where the loads are only independent of each other as long as the pump can respond fast enough to the momentary power need.

Studies into CPR systems were initiated around 1980 [1]. At that time they were mostly referred to as 'secondary controlled' systems, because the controller acted directly at the load, i.e. at the secondary side of the drive system. It was shown that rotary loads could be driven from the common pressure rail with high efficiency and with excellent controllability [1, 2, 3]. The problem at that time was that no good solution existed to drive linear loads from the rail. Throttling was out of the question, as that would spoil the efficiency of the drive line. What was needed was a transformer that could transform the pressure in the rail down to the pressures the linear loads would require, without throttling losses. Such an hydraulic transformer was developed [4, 5] but it was bulky, expensive and – due to internal losses – had an efficiency which was only barely better than that of throttling. Mainly because of the lack of a good solution for linear loads, CPR systems did not break through at that time.

In 1996, a new type of hydraulic transformer was developed [6], the 'Innas Hydraulic Transformer' (IHT), with which the full potential of the CPR system type may finally be opened up. After the IHT principle had been successfully demonstrated in a first prototype, it was developed further in the subsequent years [7, 8]. Until the end of 2001, all developments concerned adapted conventional axial piston pumps or motors, like the one shown in figure 2. At the end of 2001, it was realized that it would be necessary to design an IHT from scratch, should the CPR really be a better solution than LS. The main requirement for the dedicated IHT was a significantly larger number of pistons, to be reached without increasing the production costs.

The development of the dedicated IHT quickly resulted in a new axial piston displacement principle, which was dubbed the 'Floating Cup' (FC) principle [9]. Rotating groups based on the FC principle can be produced using chip-less production techniques. These are already common for mass produced parts like bearings but new to the production of conventional axial piston units, where machine shop production techniques are used. With these mass-production techniques, an FC rotating group with a far larger number of pistons, can be produced cheaper than a conventional rotating group.

The most important functional effects of the larger number of pistons are the reduced pulsations – very important for the IHT – and the improved low speed behavior. Simulations have shown that with an 18 piston FC IHT, the lowest stable speed can be around 2% of the maximum speed. In other words, a FC IHT that has been laid out for a cylinder with a maximum speed of 0.5 m/s, will be able to sustain a lowest cylinder speed of 10 mm/s.

It was soon realized that the advantages of the FC principle are not limited to IHTs only. It is equally interesting for hydrostatic pumps and motors and as these provide a more short-term potential, the development and prototyping were quickly redirected from the FC IHT to the FC pump and motor. Toward the end of 2002 a very satisfying pump design was reached and the development of the first FC IHT prototype was continued.

With a commercially viable FC IHT so near, studies into its application in the CPR have been intensified. Hydraulic circuit layouts were made and IHT control strategies are being developed.
The hydraulic circuit layouts, i.e. the ways in which an IHT can be connected between the CPR and a cylinder, are the subject of this article. Four of them are described and compared in chapter 3. But first, the basics of the operation of the IHT, as far as they are important for the understanding of the layout options, are treated in chapter 2. The design of the FC IHT is not treated here, as it is still in full progress. Recent information on the development of the FC displacement principle for pumps and motors, can be found in [10].

2 BASICS OF THE IHT

The first IHT prototypes were modified bent axis hydraulic motors. In order to turn these into an IHT, the standard, fixed port plate with two kidneys was exchanged for a port plate with three kidneys, that can be swivelled over a limited angle. The kidneys connect to the supply (A), the make-up (T) and the load (B) pressure respectively. The end cap was changed to accommodate these three connections. The construction of the swiveling port plate can be seen in figure 2, as well as the symbol that was coined for the IHT.

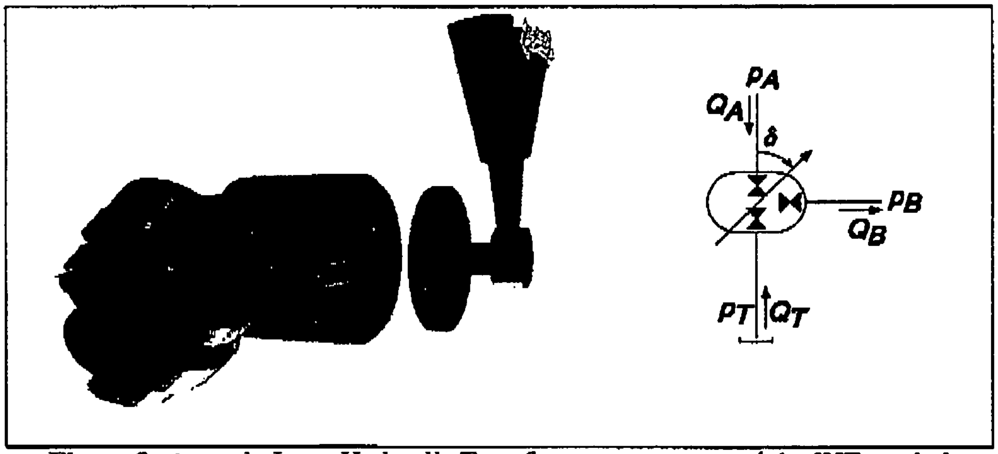

Figure 2: An early Innas Hydraulic Transformer prototype and the IHT symbol

When the barrel rotates, the cylinders pass the three kidneys sequentially. Depending on the pressure in each kidney and the cylinders angular position relative to angle at which its piston is in top dead center (TDC) position, the pistons contribute to the torque on the rotary group. It is important to notice that these hydraulic torques are the only torques acting on the IHT, it does not exchange mechanical power with its surroundings.

Assuming a constant rotational speed, the average contribution of any kidney to the torque on the rotary group, can be expressed by the general equation:

$$T_{k_{av}} = \frac{p_k \cdot V_{IHT}}{2 \cdot \pi} \cdot \left(\sin \frac{\phi_k}{2} \cdot \sin \phi_c \right) \qquad (1)$$

In which V_{IHT} is the displacement of the base unit, p_k the pressure in the kidney, ϕ_k the nominal arc length of the kidney and ϕ_c the angular position of the center of the kidney relative to TDC angle.

A lossless IHT will run at a stationary speed when the sum of the three kidney torque contributions is zero:

$$p_A \cdot \sin\frac{\alpha}{2} \cdot \sin\delta + p_T \cdot \sin\frac{\gamma}{2} \cdot \sin\left(\delta + \frac{\alpha}{2} + \frac{\gamma}{2}\right) + p_B \cdot \sin\frac{\beta}{2} \cdot \sin\left(\delta - \frac{\alpha}{2} - \frac{\beta}{2}\right) = 0 \quad (2)$$

In this equation δ is the port plate control angle and α, β and γ are the arc length of the supply (A) port, the load (B) port and the make-up (T) port.

From this equation a pressure transformation factor Π can be derived:

$$\Pi = \frac{p_B}{p_A} = \frac{-\sin\frac{\alpha}{2} \cdot \sin\delta - \frac{p_T}{p_A} \cdot \sin\frac{\gamma}{2} \cdot \sin\left(\delta + \frac{\alpha}{2} + \frac{\gamma}{2}\right)}{\sin\frac{\beta}{2} \cdot \sin\left(\delta - \frac{\alpha}{2} - \frac{\beta}{2}\right)} \quad (3)$$

This equation has been plotted in figure 3 for a lossless IHT, with three kidneys spanning an arc of 120° each and with the make-up pressure put to zero. The figure shows that an IHT can transform the rail pressure down to any value below the rail pressure but it can also amplify the rail pressure to a higher pressure.

If, at a given port plate angle, the pressures at the ports do not correspond to the theoretical transformation factor Π, the sum of the torque contributions of the three kidneys is not zero. The resulting torque will accelerate (or decelerate) the IHT, which will send more (or less) fluid to the load. As a result, the load pressure will increase (or decrease). This will continue until the theoretical transformation factor has been reached again. As the torque is large and the inertia of the IHT is low, the speed changes will occur very fast. Functionally the IHT should be thought of as a component that tends to quickly realize the load pressure level that is determined by the current supply and make-up pressure level, in combination with the port plate angle. The load will react to that pressure level with a speed that depends on the actual load process. The IHT will closely follow the load speed.
In this 'open loop' operation, the IHT can be used to control a load pressure. Mobile machines, as described before, require load speed control rather than load pressure

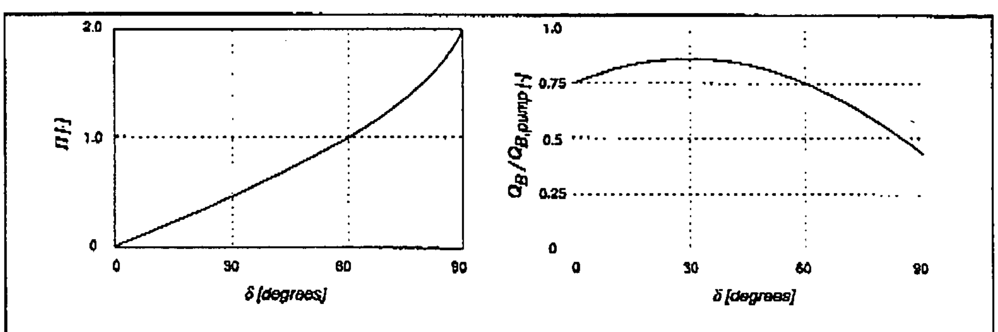

Figure 3: Transformation factor and scaled load flow

control. For speed control, the control loop has to be closed. This requires extra sensors and control hardware. On the other hand, it introduces the control quality typical for secondary control.

As as been described in [6], a fast and stable secondary controller requires feed back of both the load speed and the port plate position. As cylinder or linkage position sensors are fairly expensive and not yet common in mobile machinery, not the load speed will be controlled but the transformer's load flow. This flow can be calculated using the general expression for the average flow from the kidney of a transformer into its corresponding connection:

$$Q_{k_{av}} = \frac{\omega \cdot V_{IHT}}{2 \cdot \pi} \cdot \sin\frac{\phi_k}{2} \cdot \sin\phi_c \tag{4}$$

For the load kidney this yields:

$$Q_{B_{av}} = \frac{\omega \cdot V_{IHT}}{2 \cdot \pi} \cdot \sin\frac{\beta}{2} \cdot \sin\left(\delta - \frac{\alpha}{2} - \frac{\beta}{2}\right) \tag{5}$$

With the geometry of the IHT fixed, $Q_{B_{av}}$ is a function of the rotational speed ω and the port plate control angle δ. For an IHT with three 120^o kidneys, the function is plotted in figure 3. It has been made dimensionless by dividing $Q_{B_{av}}$ by the average flow of a pump with the same swept volume V_{IHT}, running at the same speed ω.

Using equation 5, a controller can calculate the output flow from these two internally sensed signals. The output flow controller can be integrated into the IHT structure. In this way a rugged, closed package results, which can be positioned in the harsh environment close to a cylinder. Cylinder and controller could also be integrated into one component: an intelligent hydraulic cylinder module.

The required output flows have to be communicated to the integrated output flow controller, preferably through a CAN bus. They can be generated in two ways:

- They may come directly from the joysticks, in which case the functionality of the LS system type can be exactly reproduced, with the added advantage that the sensitivity of the joysticks can easily be adjusted to the drivers liking or to the job at hand.
- They may be generated by a global vehicle controller, aiding the driver with complex synchronization tasks. If the global controller fails, the local controllers can fall back to reacting directly to the joystick signals. If a global controller is present, it is likely that also cylinder or linkage position sensors are available. These can be used to watch the performance of the local output flow controllers and compensate for temperature and wear effects. This can either be left to the local controllers, taking the actual position information from the CAN bus, or it may be left to the global controller. If the global controller detects a mismatch, it can adjust the required flow signals to the local controllers. In an alternative set-up it can send information that allows the local controller to adjust its control parameters. The latter is preferred, as it enables a 'learning' compensation algorithm.

3 POSSIBLE CIRCUIT SOLUTIONS FOR IHT BASED CYLINDER CONTROL

Most implement functions in mobile machinery are driven by differential cylinders that can operate in four quadrants: they have to be able to exert forces in extending and in retracting direction, independent of the direction in which they are moving. Magnitude and direction of the forces can change suddenly, for instance when the bucket of an excavator is lowered to the ground to start digging. Even when the bucket hits the ground and the force suddenly changes direction, the movement of the cylinder controlling this process should be as continuous as possible.

There are several ways in which an IHT can be used to realize four quadrant operation of a differential cylinder connected to a Common Pressure Rail. In this chapter, four fundamentally different alternatives will be introduced and compared.

In mobile machines which are also used for lifting loads, a cylinder lock-up function is often required. This function ensures that a load does not lower under the influence of gravity, even when it is left hanging for a prolonged period, possibly with the machine shut off. In most cases, a lock-up on one cylinder side is sufficient. In the last three alternatives presented in this chapter, the optional lock-up function has been incorporated in the scheme. It is assumed here that the gravity load tends to pressurize the bottom side of the cylinder. The lock-up function has been added to that side. Of course, if a cylinder is mounted in the vehicle in such a way that the gravity load pressurizes the ring side, the lock-up has to be at that side. In all alternatives this can be done without loss of functionality.

If a lock-up function is present, it can also be used to realize a perfect standstill of a non-commanded cylinder when it experiences gravity loads or reaction forces induced by the actions of other cylinders. This functionality will be described in more detail too.

3.1 A four quadrant IHT

The most obvious way to realize four quadrant operation would be to use the four quadrant transformer (the '4Q IHT') that has been designed for and tested in the hydrostatic wheel drive of a fork lift truck. It has been described in earlier publications [11] and is shown again in figure 4. The design requires a special port plate and a 4/2 valve. Using the same circuit lay-out, it can be used to control a differential cylinder.

The idea for the 4Q IHT originates from the symmetrical nature of the IHT: if the port plate of an IHT is turned to negative control angles, the IHT operates as described before, only its B and T port switch function. So, for a 4Q IHT it should be possible to turn the port plate to positive as well as negative control angles.

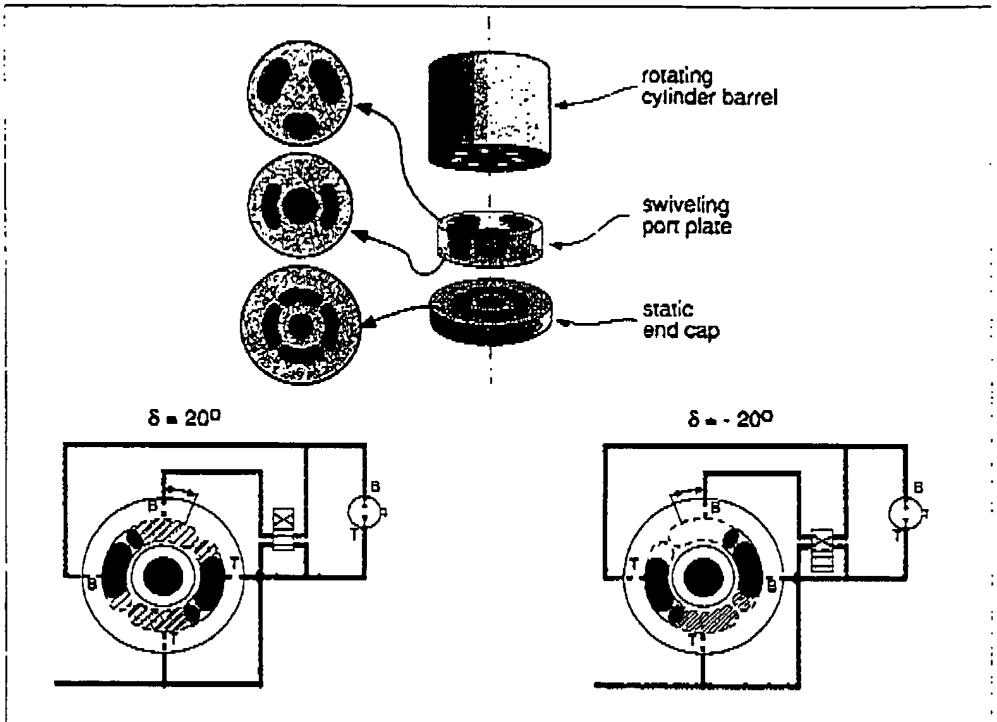

Figure 4: The port plate and switching valve of a four quadrant IHT

In order to be able to start up an IHT from zero speed at high loads and because the pressure transformation efficiency of an IHT is not 100%, even an IHT that does not have to amplify the rail pressure needs a maximum control angle well over 60°. The required maximum angle for the FC IHT design is still unsure, as no such IHT prototypes have been tested yet. Past measurements of IHTs based on conventional axial piston units, suggest a worst case start-up angle of 90°. For the four quadrant transformer this implies that it should be possible to rotate the port plate from −90 to + 90°.

This, however, introduces a problem: all three kidneys in the port plate have to connect to static ports in the end cap, while the port plate itself has to be able to rotate over 180°. If the three kidneys in this interface are arranged on the same circumference, a total of 540° should be available, which a circle cannot offer.

A solution has been found in a port plate design in which the A kidney is rerouted from its position on the front side of the port plate to a central hole in the back side. In this way more room is created for the B and T kidneys on their common circumference. By creating four ports in the end-cap, two of which are switched when the port plate rotation angle changes sign, a full rotational freedom of 90° in both directions is obtained. As has been shown in [11], a 4Q IHT needs at least a 3/2 valve because the make up

flow has to be switched when the ports switch function. With a 4/2 valve, the make up flow as well as the two 90° ports can be switched. The valve has to switch when the port plate control angle changes sign. This is a geometrically defined condition, so theoretically the valve can be incorporated in the port plate. If a separate valve is chosen, its operation can be directly linked to the port plate position. Even when the valve is switched, the flows between the IHT and both sides of the load (motor or cylinder) are uninterrupted and do not change sign. This implies that switching the force direction on the load does not cause pressure peaks or temporarily blocked movement of the cylinder.

The optional lock-up function can be realized in exactly the same way as described in the section 3.2.

In the 4Q design, the front side and back side ports of each port plate kidney are offset. This leads to large unbalanced static torques on the port plate, which have to be taken by a port plate bearing. Because this bearing has to be accommodated and because the ports have to be re-routed between front and back-side of the port plate, the port plate construction of this 4Q IHT is complex. It gets even more complex if the function of the 4/2 valve is integrated in the port plate.

3.2 A two quadrant IHT in combination with a 4/2 spool valve

The second alternative uses a normal spool valve to switch the load connection of the IHT from the bottom side of the cylinder to the ring side, while simultaneously switching the the low pressure connection of the cylinder from its ring side to its bottom side. In this way the direction of the force exerted by the cylinder is reversed while the load and low pressure ports of the IHT do not have to be switched. A two-quadrant ('2Q') IHT can be used, in which the routing of the ports in the port plate is straightforward. Figure 5 shows the circuit.

The cartridge valve in figure 5 provides the optional lock-up function. It is piloted by a small electromagnetic 3/2 valve, which is connected in such a way that the lock-up valve is closed when the system is not electrified. The lock-up valve closes the cylinders bottom side completely and thus prevents the cylinder from moving in retracting direction.

As mentioned in the introduction to this chapter, the lock-up function can also be used to prevent the cylinder from moving when another cylinder causes reaction forces. In order to realize this, the lock-up valve is closed and the IHT is controlled to the maximum pressure on the ring side of the cylinder. The cylinder is then secured between the pressures at the bottom and ring side of the cylinder.
The bottom side of the cylinder is protected by a pressure limiting valve, as this volume

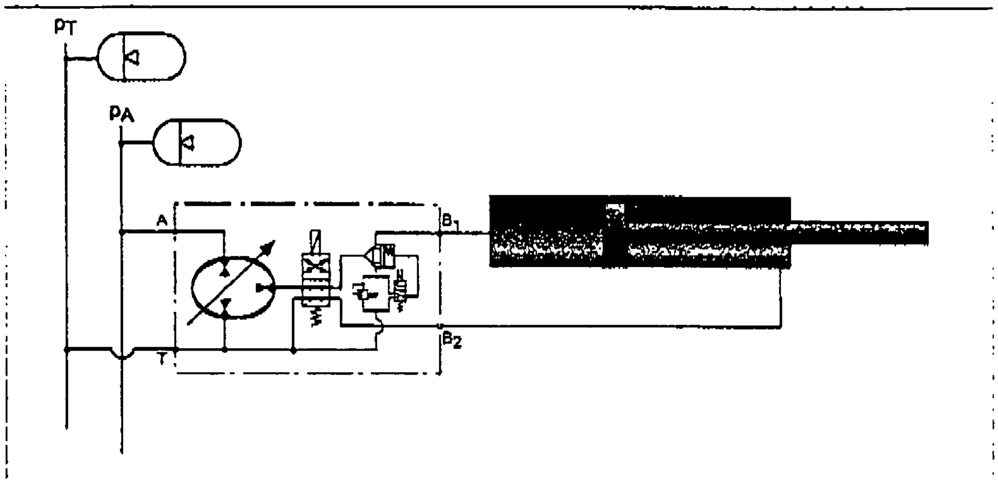

Figure 5: Control of a differential cylinder using an IHT and a directional valve.

is completely shut off from the system when the lock-up is active. On the ring side, the normal IHT behavior ensures that the pressure can never rise above the maximum level set by the transformers control angle. If, due to the action of another cylinder, the pressure at the ring side tends to rise above that value, the transformer will simply start to rotate allowing the cylinder to move and thus maximizing the pressure. Due to the movement, the pressure at the bottom side of the cylinder will drop. When it falls below the low pressure acting at the other side of the lock-up valve, this functions as a one-way valve and lets oil flow into the bottom side of the cylinder.

From a control point of view, this circuit solution has a distinct disadvantage. If the loading situation requires that the load pressure connection of the IHT is switched from one cylinder side to the other, while the cylinder has to continue moving in the same direction, the IHT has to change its direction of rotation. This means that the IHT has to break to a stop and subsequently accelerate in the opposite direction. Because the IHT's inertia is low compared to its maximum torque capability, this will happen very quickly. Nevertheless, there will be a brief moment with pressure peaks in the load connection of the transformer and a severe discontinuity in the control strategy. At the same moment, also the flow between the valve and the load is temporarily blocked off. This requires careful tuning of the transition between the two positions of the valve in order to ensure smooth switching under all circumstances.

It should be noted that in stead of a 4/2 valve, also four 2/2 cartridge valves can be used, piloted by a small spool valve. The advantages of using cartridge valves are:

- In this system, the valves realizing the directional function have to be designed to take the full IHT flow. This would lead to a large 4/2 valve. Cartridge valves could present a more compact and less costly solution.

- A lock-up of both cylinder sides can be realized by adding a third position to the pilot valve. A separate piloted cartridge valve for the lock-up function is no longer necessary, which saves costs.

3.3 A two quadrant IHT in combination with a pressurized cylinder rod side

In this third strategy, the rod side of the cylinder is always connected to the high pressure line of the CPR. The IHT is designed to have an amplification capability. When a large force in extending direction is required, the bottom side pressure is raised above the rod side pressure. The area ratio between bottom and rod side of the cylinder helps to overcome the force generated by the ring side. Figure 6 shows a sketch of this circuit option.

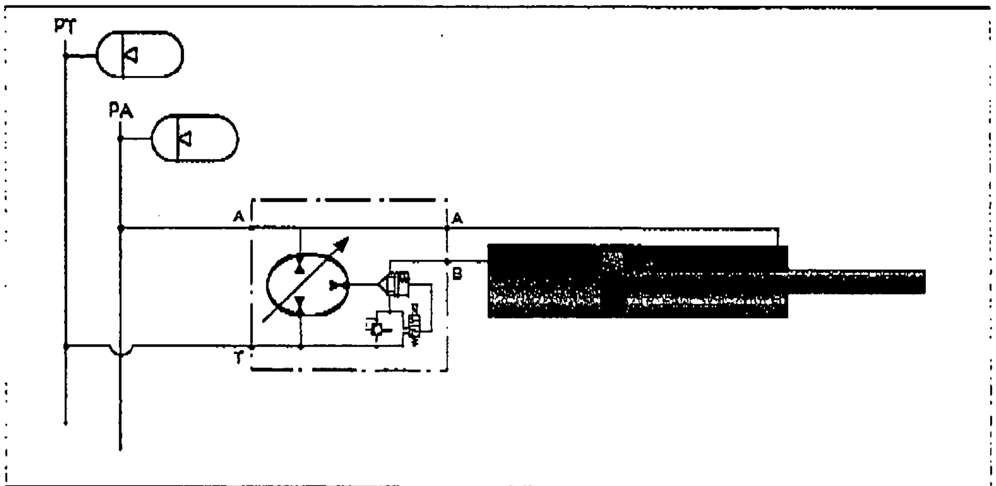

Figure 6: IHT control of a double acting cylinder with a pressurized cylinder rod side.

In this solution, the lock-up function for gravity loads and the zero speed capability can be realized in the same way as in the previous circuit options. In this case, however, the pressure that can be applied to the ring side can not be varied with the IHT. It is always equal to the high rail pressure (p_A) at the ring side of the cylinder. Again, if the extending force exceeds the lockup force, the cylinder will move.

A very significant advantage of this solution is the continuous control of the IHT. The control angle directly sets the hydraulic force of the cylinder, the IHT speed smoothly follows the resulting load flow. Neither the direction of rotation of the IHT nor the port plate control angle need to be changed suddenly.

A disadvantage of this solution is that the cylinder bottom area has to be increased, if

the same force in extending direction has to be generated. This is due to the fact that the pressure amplification ratio can not be increased at will. Above a certain value, the efficiency of the pressure transformation diminishes significantly.
Measurements have shown that the efficiency of an IHT running at maximum speed at a port plate control angle of 90°, is aproximatley 70%. So:

$$\Pi_{actual_{(90,n_{max})}} = \eta_{\Pi_{(90,n_{max})}} \times \Pi_{theoretical_{(90,n_{max})}} = 0.7 \times 2 = 1.4 \tag{6}$$

Assuming that the CPR supply pump and the LS pump have the same maximum pressure capability of 30 MPa, the maximum output pressure of the IHT is thus 1.4 × 30 MPa = 42 MPa. At the same input pressure of 30 MPa, the maximum output pressure of an LS block is about 28 MPa. As mentioned in chapter 1, this pressure drop over the valve block is essential to the functioning of the LS system.
For a typical differential cylinder, the ratio between the rod-side area and the bottom side area is 1.5. With these data, the required bottom size area can be calculated from the requirement that the extending force in the IHT based system is equal to that in the LS system:

$$F_{extension_{IHT}} = F_{extension_{LS}} \tag{7}$$

This yields the following equation:

$$42 \times A_{bottom_{IHT}} - 30 \times A_{ring_{IHT}} = 28 \times A_{bottom_{LS}} \tag{8}$$

Assuming that the retracting forces in both systems should be equal:

$$\begin{aligned} 42 \times A_{bottom_{IHT}} - 28 \times A_{ring_{LS}} &= 28 \times A_{bottom_{LS}} && (9)\\ 42 \times A_{bottom_{IHT}} - 28 \times \frac{A_{bottom_{LS}}}{1.5} &= 28 \times A_{bottom_{LS}} && (10)\\ & && (11) \end{aligned}$$

which gives the relationship between the bottom side area of the cylinder in the IHT based system to that in the LS system:

$$A_{bottom_{IHT}} = 1.11 \times A_{bottom_{LS}} \tag{12}$$

This means that in this system lay-out, the inner cylinder diameter has to be increased by about 5.4%, in order to achieve the same cylinder force potential as the LS system.

Figure 3 shows that if the port plate is governed in the amplification range, the load flow per revolution quickly diminishes with the port plate angle. As a result, the swept volume of an IHT that has to deliver the desired flow at large amplification angles, has to be larger than the swept volume of an IHT that does not need amplification for its operation. This effect is augmented somewhat because the maximum speed of any axial piston unit decreases with increasing swept volume. Consequently, this strategy does not only require a larger cylinder but also a significantly larger IHT.

3.4 A two quadrant IHT in combination with a switched cylinder rod side

In the last strategy, the rod side of the cylinder is connected to the high pressure line when a force in retracting direction or a moderate force in extending direction is required. When a large force in extending direction is necessary, the rod side is switched to the low pressure line. The solution is sketched in figure 7. Here, the IHT does not

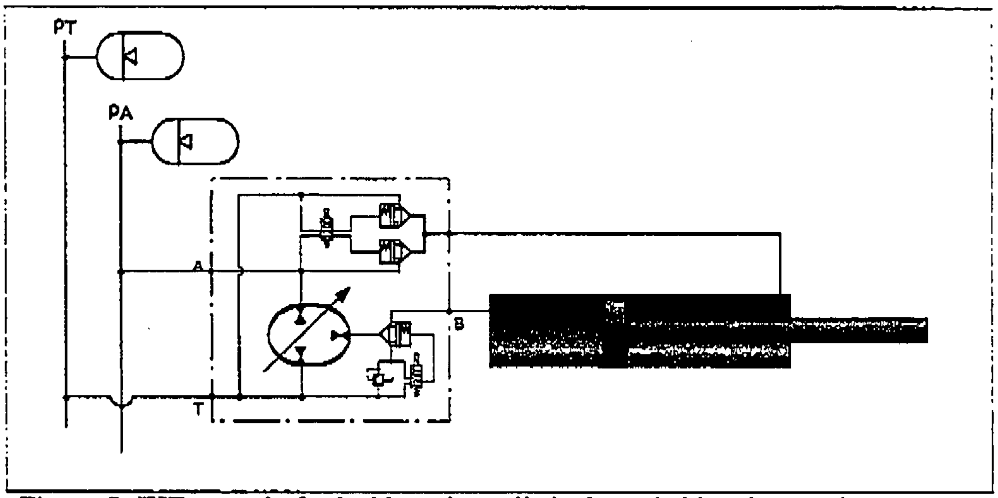

Figure 7: IHT control of a double acting cylinder by switching the rod side pressure.

have to go deeply into the amplification range to be able to realize the same cylinder forces as the conventional system type. The largest part of the amplification range can be used to start up the IHT. The load flow per revolution is lower, but at low speeds that is not important.

For the lock-up and stand still functionality – here also incorported through a piloted cartridge valve – the same considerations are valid as for the circuit option of section 3.3.

The drawback for this circuit option are the extra valves, which are necessary at the rod side. Compared to the solution of section 3.3, they add extra costs. In the scheme presented in figure 7, two inexpensive cartridge valves have been used, piloted in tandem by a small 4/2 valve. With this setup, unlimited pressure peaks will not occur if the switchover is made when the load is still moving, as the cartridge valves will act as one-way valves to the CPR system.

The port plate control is not continuous anymore. At some point the IHT controller has to make the decision to switch the cartridge valves. It is not difficult to incorparate this decision in the controller. When the controller detects that the realised cylinder speed differs from the requested speed, it will respond, according to the continuous control

algorithm, by first adjusting the port plate control angle. It should switch the cartridge valves when it detects that:

- the required speed cannot be met,
- the port plate angle is approaching its limit value and
- the position of the 4/2 valve is such that switching it can increase the force in the right direction.

When the controller would just switch the valve, the cylinder force would change stepwise. The magnitude of this force change can be predicted, so it is possible for the controller to adjust the port plate angle, also stepwise, in order to smoothen the switch.

4 Conclusion and outlook

With the rather qualitative evaluation of the four circuit solutions presented in chapter 3, it was concluded that the option in which the cylinder ring side is switched between supply and make-up pressure is the most promising one. It will be used in the 'IBIS' project, a European project in which the boom functions of a medium sized excavator will be changed to IHT control. A controller for the combination of the FC IHT with this circuit option, is being developed.

The size of the FC IHT prototype under development is suited for the excavator. The prototype will be used for the first functional testing of the new FC principle in an IHT. It will also be used for the excavator project, as as a pre-prototype with which the circuit and the control algorithms can be tested before they are incorporated in the excavator.

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